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TECHNICAL NOTE
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ON THE EMPIRICAL CORRELATION OF CRITICAL BOILING HEAT FLUX FOR CHANNELS OF VARIOUS CROSS SECTIONS AND ORIENTATION

By Uwe H. von Glahn

Lewis Research Center
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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
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SUMMARY

A correlation study is made of some of the experimental maximum heat-flux data (burnout condition) with net vapor generation available in the literature. The data studied herein were obtained with water flow through circular tubes, rectangular channels, and annular tubes. Vertically upward, horizontal, and 45° inclined flow directions are considered. All channels were uniformly heated in the direction of flow. Correlation is achieved through modification of an existing empirical relation by the use of an additional channel-shape factor. The empirical relation itself consists of an enthalpy ratio and a function consisting of several dimensionless groups containing variables that include mass velocity, channel geometry, and fluid property terms. All fluid property terms are evaluated at the fluid saturation temperature associated with the burnout condition.

INTRODUCTION

In the design considerations of boiling fluid devices to be used for the generation of large amounts of vapor, the best efficiency in terms of large heat fluxes and low wall-to-fluid bulk temperature differences can be achieved within the nucleate-boiling regime. Transition from the nucleate- to the film-boiling regime, with its accompanying high wall-to-fluid bulk temperature differences, however, can result in the destructive failure of the component. The heat flux associated with the transition from nucleate-bulk to film boiling is herein called the critical heat flux. For a flowing system this critical heat flux can be defined as the flux immediately before the transition from a high heat-transfer coefficient to a lower value at some unspecified location along the channel or tube axis. The point along the channel surface at which this transition in heat-transfer coefficient occurs is frequently called the burnout point because of the possibility of material failure. The distance measured along the tube or component axis from the inlet to the burnout location is designated herein as the critical length.

An empirical correlation of the critical heat flux for conditions of net vapor generation in forced flow upward through uniformly heated circular tubes is presented in reference 1. Correlation was achieved for water, liquid hydrogen, and liquid nitrogen through the development of an empirical relation involving an enthalpy ratio and a function consisting of several dimensionless groups. These dimensionless groups contain variables that include mass velocity, tube geometry (diameter and length), and fluid property terms. The latter terms were evaluated at the fluid saturation temperature associated with the fluid pressure at the burnout location.

In the present study, conducted at the NASA Lewis Research Center, the empirical relations for vertical upward flow through circular tubes (developed in ref. 1) were modified and applied to data that included variations in channel cross section and channel inclination or orientation. The extended correlation presented herein is based on sparse and incomplete ranges of data that include the following tube or channel characteristics: (1) vertical upward flow through rectangular channels and annular and circular tubes, (2) 45° inclined upward flow through rectangular channels and circular tubes, and (3) horizontal flow through annular and circular tubes. The data used for the present correlation are for uniformly heated tubes and channels with water as the working fluid and cover a saturation pressure at the burnout location that ranges from approximately 2 to 136 atmospheres.

DATA ANALYSIS AND PROCEDURE

The procedures used in manipulating the data for correlation purposes consisted of a series of trial-and-error solutions. In these solutions, the empirical correlating relations of reference 1 were modified by the application to the data of successively selected dimensionless channel-shape parameters. The objective in all such applications was to normalize as much of the noncircular channel critical-heat-flux data as possible on a single curve with that for circular tubes. Underlying this objective was the criterion that the correlation presented in reference 1 should be considered the foundation or base upon which to build and should in no way be affected by the applied channel-shape parameters. Furthermore, any additional parameters or modified exponents to parameters should not yield unreasonable or unrealistic values at their limits.

The average curve and the ±15-percent scatter-band limit curves of reference 1 (containing about 85 percent of the data points shown in ref. 1) serve as the basis of comparison for the noncircular channel data. Because of the scarcity of data available for noncircular channels, all data points given in the reference reports (except as noted) are used herein, and no data selection similar to that employed in reference 1 was made. As in reference 1, data with an entering quality at the entrance to the heated channel were not included in the present work.

In table I are listed the total number of critical-heat-flux data points associated with net vapor generation in the references used herein (refs. 2 to 9) and the range of variables covered in the references cited. It should be noted that for the water data considered herein the burnout location always occurred at the tube exit, and the pressure noted in table I is that measured at or near the burnout location.

DEVELOPMENT OF MODIFIED CORRELATION PARAMETERS

Channel Geometry Considerations for Vertical Upward Flow

In reference 1 an empirical relation was developed for circular tubes with vertical orientation relating a critical vaporization parameter X_C to a parameter that combined flow considerations, tube characteristics, and fluid properties. The relation developed was expressed as

$$X_C = f\left[\underbrace{\frac{GD}{\left(\frac{L}{D}\right)^{1.3}}}_{\substack{\text{flow}\\ \text{parameter}}} \quad \underbrace{\frac{Pr_v^{0.4}}{\mu_v}\left(\frac{\rho_l - \rho_v}{\rho_v}\right)^{0.4}\left(\frac{\mu_v}{\mu_l}\right)^{1.7}}_{\text{fluid property parameters}} \quad N_B\right]_s \tag{1}$$

where

$$X_C \equiv \frac{\frac{Q_C}{W}}{\Delta h} = \frac{4q_C L}{GD\ \Delta h} \tag{2}$$

(All symbols are defined in the appendix.)

In applying equations (1) and (2) to noncircular channels, the diameter D is replaced by the equivalent diameter D_e. (The equivalent diameter is defined as four times the channel cross-sectional area divided by the perimeter of the heated surface.) The use of D_e in equations (1) and (2) to put the annular tube and rectangular channel data on the same basis as the circular tube data resulted in a divergence of the data. The divergence was evidenced by values of X_C higher than the limiting value of 1.0 for the circular tubes and by a variation of the noncircular channel data with the ratio L/D_e.

Analysis of the rectangular and annular data showed, however, that with the use of the defined equivalent diameter and a properly selected channel-shape factor these data could be normalized on a single curve together with the circular tube data. A satisfactory channel-shape factor F, that depended on the ratio of an effective channel height to the equivalent diameter, and in the case of an annular tube, an additional factor consisting of the ratio of the tube diameters was determined by trial and error.

The effective channel height b_e for channels in which the fluid is enclosed or nearly enclosed by the heated perimeter (i.e., circular tubes or large aspect ratio rectangular channels with the two large parallel sides heated) was found to be a function of the radius or one-half the channel height, respectively. For channels in which the fluid is only partly enclosed by the heated perimeter (i.e., annular tubes with inner tube perimeter heated and rectangular channels with only one side heated), the effective channel height was found to be a function of the entire channel height above the heated surface.

The channel-shape factors for circular and rectangular channels can be written as

$$F \equiv \frac{2b_e}{D_e}$$

and those for annular tubes

$$F \equiv \left(\frac{2b_e}{D_e}\right)^{\left(\frac{D_1}{D_2}\right)^{0.4}}$$

The channel-shape factors for the configurations considered herein are summarized in table II.

It should be observed that as the ratio of D_1/D_2 in the exponent of the annular-tube shape factor approaches 1.0, that is, as D_1 approaches D_2, the value of F approaches 0.5 or that of a rectangular channel. Also, for an annular tube having only the outer tube heated, F might be expressed as

$$F = \left(\frac{D_2}{D_2 + D_1}\right)^{\left(\frac{D_1}{D_2}\right)^{0.4}}$$

As the exponent D_1/D_2 for the aforementioned configuration approaches 1.0, F again approaches 0.5; however, as D_1/D_2 approaches 0, F approaches 1.0, the circular tube value for F. In an annular tube configuration with both tubes heated, F might be expressed by $0.5^{(D_1/D_2)^{0.4}}$, which yields values of F that range from 0.5 to 1.0 depending on the ratio D_1/D_2.

Details of the use of the channel-shape factors for data correlation are discussed in the following sections.

<u>Modification of critical vaporization parameter</u>. - Examination of the data obtained with rectangular channels (ref. 2) in the high fluid quality range consistently shows values of X_c exceeding 1.0 (the maximum value for circular tubes) by as much as 35 percent when the equivalent diameter only was used in equation (2). Normalization of all the data with respect to X_c (i.e., X_c not exceeding a value of 1.0) was obtained by the addition of an appropriate channel-shape factor F in equation (2) such that a more general form of X_c could be written as

$$X_c \equiv \frac{(Q_c/W)F^{0.5}}{\Delta h} = \frac{4q_cL}{GD_e\ \Delta h}\ F^{0.5} \qquad (3)$$

Modification of flow parameter $GD/(L/D)^{1.3}$. - As in the case of the critical vaporization parameter, the circular tube diameter D in the flow parameter portion of equation (1) was replaced by the equivalent diameter D_e for other channel cross sections. For the rectangular and annular tube data, the exponent of the ratio L/D_e, as well as the flow parameter itself, appeared to be functions of the channel-shape factor. The noncircular channel data could be correlated with circular tube data through the use of the term

$$\frac{F^{4.5}}{\left(\frac{L}{D_e}\right)^{1.3(F)^{1.5}}} \tag{4}$$

in place of the term $(L/D)^{1.3}$ in equation (1). For the rectangular channels used herein, the exponent for L/D_e is a constant equal to 0.46, and the value of $F^{4.5}$ is also a constant equal to 0.04415. (For circular tubes these values are 1.3 and 1.0, respectively.) For annular tubes these functions are variables depending on the tube-diameter ratio term in F, as discussed previously.

The complete correlation equation for the determination of the critical heat flux with vertical upward flow through channels is expressed by the following relation:

$$X_C = f\left[\frac{GD_e Pr_v^{0.4}}{\mu_v} \frac{F^{4.5}}{\left(\frac{L}{D_e}\right)^{1.3(F)^{1.5}}} \left(\frac{\mu_v}{\mu_l}\right)^{1.7} \left(\frac{\rho_l - \rho_v}{\rho_v}\right)^{0.4} N_B\right]_s \tag{5}$$

where X_C is defined by equation (3).

Correlation of rectangular channel data. - The correlation of critical heat-flux data for boiling water in terms of equations (3) and (5) for vertical upward flow through rectangular channels is shown in figure 1. Also shown are the average curve and the scatter-band limit curves obtained with the circular tube data of reference 1.

The data shown in figure 1(a) and some additional classified data (not shown herein) were used to establish the channel-shape factor and the necessary exponents for rectangular channels. Of the correlated data shown, 80 percent are within the ±15-percent scatter-band limit curves given in reference 1 for the circular tube data.

The rectangular channel data shown in figure 1(b) (refs. 3 and 6) constitute an independent check on the development of the channel-shape factor because (1) the data were not used in establishing the channel-shape factor, (2) the data were obtained from other sources, and (3) the data cover a range of larger channel geometries than those used in establishing the channel-shape factor. Good correlation of these data with the circular tube data (as well as those of the other channels shown (fig. 1(a)) indicates that the channel-shape factor can be used with a good degree of confidence to extend the correlation beyond the ranges specifically covered by the available data.

Correlation of annular tube data. - The correlation of critical boiling heat-flux data in terms of equations (3) and (5) for vertical flow upward through annular tubes (refs. 5 to 7) is shown in figure 2(a). The average and scatter-band limit curves obtained with circular tube data are again shown for purposes of comparison. While the annular tube data generally lie near the circular tube data, a significant and consistent trend with the amount of liquid subcooling at the tube entrance is exhibited in the data. In all cases the highest values of X_C at a given flow parameter value (abscissa in fig. 2(a)) are associated with the largest liquid subcooling at the tube entrance, the converse also being generally true. No such effects due to liquid subcooling at the channel entrance were obtained with either circular or rectangular channel configurations. In terms of the parameters in equations (3) and (5), an effect of liquid subcooling at the tube entrance is usually associated with local-boiling critical-heat-flux values rather than with bulk-boiling critical-heat-flux values. On the other hand, the total heat input per pound of flowing liquid exceeds the enthalpy of the subcooled liquid, and on this basis it could be reasoned that these data constitute bulk-boiling critical-heat-flux values. Because the bulk temperature at the burnout location is not available for the data shown in figure 2(a), it cannot be established with complete certainty whether or not these data are local- or bulk-boiling critical-heat-flux values.

An explanation for the data trend with the amount of subcooling can perhaps be obtained from a consideration of the two-phase flow inside annuli such as that used to obtain the data shown in figure 2(a). In an annulus, a progressively larger volume of cool fluid must be heated with increasing radial distance from the heated inner tube compared with that for a circular tube or a rectangular channel. As a consequence of this radial increase in volume of cool fluid, a fluid temperature-mixing problem may exist for flow through an annular tube that does not exist for the other configurations.

In an effort to correlate these same annular tube data and those for circular tubes and rectangular channels with an analytical solution, it was found in reference 4 that a change in the value of the constant from 1.0 for the two channel geometries to 6.5 for the annular tubes was required. Reference 4 suggests that the change in the constant required in order for the equation to fit the data "is due primarily to retention of a relatively larger amount of the available liquid in the film on the unheated surfaces at the expense of the effective liquid concentration in the channel core at the critical heat flux condition." This hypothesis, according to reference 4, is substantiated by the fact that "annular test section data taken with a ribbed outer wall designed to minimize attachment of the liquid on the unheated surface are in good agreement with calculations" using a value of 1.0 for the constant in the equation developed in reference 4 and by motion pictures that show "a relatively thicker, more placid layer of liquid flowing up the unheated wall, compared to the thinner, more agitated liquid film on the heated wall".

A plot of the annular tube data taken with a ribbed outer wall in the annulus (ref. 4) is shown in figures 2(b) and (c). In figure 2(b), these data are shown in terms of equations (3) and (5). In general, the X_C values are higher than those given by the average curve for the circular tube data by about 25 percent. This increase in X_C is attributed to the generally increased turbulence level of the two-phase fluid flowing through the annulus; turbulence has the effect of increasing the boiling heat-transfer coefficient, which results in the obtaining of higher critical-heat-flux values. Of more specific interest is the effect of this turbulence or better thermal mixing on the variation of X_C with liquid subcooling enthalpy. This variation together with that obtained by using a smooth outer wall in the annulus is shown in figure 2(c). For similar ranges of liquid subcooling enthalpy, the ribbed outer wall reduced the X_C variation to 12 percent from about 30 percent for the smooth outer wall.

Further documented evidence must be obtained with annular tube configurations in order to determine whether the data shown in figure 2(a) are local- or bulk-boiling critical-heat-flux values and thereby determine whether additional correlation terms are required to account for liquid subcooling trends that may occur in some channel geometries.

Channel Orientation Considerations

At present, the effect of channel orientation on the critical heat flux for a forced flow system is difficult to evaluate because of the great lack of data. It can be postulated that the change in channel orientation from vertical to horizontal with circular tubes and rectangular channels could result in an increased agglomeration of vapor bubbles on the heated upper surface of the horizontal channel. In this event, a burnout condition would be reached at a lower critical heat flux for otherwise comparable flow conditions. For an annular tube with the inner tube heated, however, the result could be somewhat different: that is, the bubbles would tend to agglomerate on the upper portions of the unheated tube perimeter, and thus burnout could possibly be delayed.

Data in terms of equations (3) and (5) for a channel orientation of 45° inclined upward flow (ref. 4) and horizontal flow (refs. 8 and 9) are shown in figures 3(a) and (b), respectively. Also shown in figure 3 are the average and scatter-band limit curves for vertical upward flow through circular tubes. With a channel inclination of 45°, the sparce data for both the circular tubes and rectangular channels are represented reasonably well by equations (3) and (5). With horizontal flow through annular tubes (fig. 3(b)), however, the data deviate greatly from the correlation given by equations (3) and (5). Data from reference 8 are generally significantly below the curves for vertical upward flow data, while data from reference 9 are for the most part significantly higher than these curves. Two preliminary data points (unpublished NASA data) with a horizontally oriented circular tube indicated reasonable agreement with the correlation given by equations (3) and (5). A data point obtained in this same study with a circular tube oriented vertically is also shown for comparison purposes. No apparent effect due to orientation for circular tubes appears evident from these meager data.

Studies of vertical upward flow through tubes and channels suggest that the data of reference 8 perhaps should not be considered representative of maximum critical-heat-flux values, but rather of lower values associated with an unstable flow condition of unknown magnitude and origin with consequent submaximum critical-heat-flux values. Examination of the data, however, could also suggest that the exponents in equation (4) are a function of the orientation angle. Furthermore, examination of the data from reference 9 indicates that the Q_C/W in general does not greatly exceed the liquid subcooling enthalpy at the channel entrance. Thus, as discussed for figure 2(a), these data are suspect in that they may be local-boiling critical-heat-flux values rather than values associated with net vapor generation.

In view of the scarcity of critical-heat-flux data available for configurations with a horizontal orientation, consideration of an orientation function in equations (4) and (5) does not appear to be warranted at this time.

CONCLUDING REMARKS

Some of the problems encountered in correlating presently published critical-heat-flux data are perhaps worth emphasizing to the current researcher in order to obtain a greater degree of excellence in future two-phase data. All too often, apparent trends in the data are due to the particular research loop and instrumentation used by individual researchers to establish critical-heat-flux values. Particular attention should be exercised to avoid inadequacies and incompleteness in data measurements and criteria (in particular, the techniques and judgement used to define the often nebulous criteria of flow instability and its effect on two-phase heat transfer) used to establish critical-heat-flux values.

It is also to be expected that critical-heat-flux values obtained in the future with facilities incorporating automatic burnout detectors that sense the large wall-temperature excursions associated with the critical heat flux may be higher than those reported by previous researchers. Based on limited current research results, these increases in critical heat flux at critical vaporization parameters of less than 0.7 may amount to as much as 25 percent compared with "manual" operation of the research equipment. The unpublished NASA data were obtained by using an automatic burnout detector such as mentioned above and the resultant values of critical vaporization parameters are considered representative of the higher critical-heat-flux values that will be reported by researchers using such devices.

SUMMARY OF RESULTS

The results from a study of the critical heat flux for channels of various cross sections and orientation can be summarized as follows:

1. Parameters have been evolved for boiling heat-transfer data obtained with water that correlate the critical boiling heat flux for forced flow upward with net vapor generation over a wide range of operating conditions and several

channel or tube configurations. The empirical relations developed herein are based on a previous correlation (NASA TN D-1285) that was applicable for various fluids, including liquid hydrogen, liquid nitrogen, and water.

2. Limited data indicate that the correlation may also be applicable to inclined and horizontal channel orientations; however, in view of the scarcity of information available for channels with orientations other than vertical, as well as large variations in the results obtained by different investigators, further well-documented, systematically obtained data are required to verify the terms used in the present correlation for relating channel geometry and orientation.

Lewis Research Center
National Aeronautics and Space Administration
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APPENDIX - SYMBOLS

A	cross-sectional flow area of channel or tube, sq ft
b	channel height, ft
b_e	effective channel height, ft
D	tube diameter (I.D.), ft
D_e	equivalent diameter, $4A/p_h$, ft
F	channel-shape factor, dimensionless
f	functional notation
G	mass velocity (flow rate per unit cross-sectional area of tube), lb mass/(hr)(sq ft)
g	acceleration (gravity), 4.17×10^8 ft/hr^2
g_c	conversion constant between (lb force) and (lb mass), 4.17×10^8, (lb mass/lb force)(ft/hr^2)
h	fluid enthalpy, Btu/lb mass
Δh	total enthalpy change required for complete vaporization of incoming fluid, $\cong h_v - (h_l)_{t_i}$, Btu/lb mass
L	critical length from tube or channel entrance to burnout point, ft
N_B	boiling number (fluid property parameter), $\dfrac{\mu_v^2 \sqrt{(\rho_l - \rho_v)g}}{\rho_v (g_c \sigma_L)^{1.5}}$, dimensionless
Pr	Prandtl number, dimensionless
p_h	heated perimeter, ft
Q	total heat input from tube entrance to burnout point, Btu/hr
q	heat flux per unit surface area, Btu/(hr)(sq ft)
r	radius, in.
t	static temperature, $^{\circ}$F
W	weight flow rate of fluid, $G(\pi D^2/4)$, lb mass/hr

w channel width, ft

X_C critical vaporization parameter, dimensionless

μ fluid viscosity, lb mass/(hr)(ft)

ρ fluid density, lb mass/cu ft

σ_L fluid surface tension, lb force/ft

Subscripts:

C critical value of heat flux (burnout condition)

l liquid

s saturation

t_i inlet temperature

v vapor

1 heated inside diameter (annular tube)

2 outside diameter (annular tube)

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TABLE I. - DATA SELECTION AND RANGE OF VARIABLES USED FOR CORRELATION PURPOSES

Reference	Orientation	Nominal fluid pressure at burnout location, lb/sq in. abs	Nominal tube or channel dimensions, in.			Nominal critical channel length to equivalent diameter ratio, L/D_e	Nominal fluid subcooling at inlet, $^{\circ}F$	Mass velocity, G, $\frac{\text{lb mass}}{(\text{hr})(\text{sq ft})}$	Heat flux, q, $\frac{\text{Btu}}{(\text{hr})(\text{sq ft})}$	Total number of data points with net vapor generation
			b	w	L					
2	Vertical	800 to 2000	0.101	1.0	6	30	12 to 537	$0.03 \text{ to } 0.78\times10^{6}$	$0.21 \text{ to } 2.25\times10^{6}$	58
		2000	.097		12.062	62	8 to 555	.02 to 2.92	.12 to 1.24	93
		2000	.050		12.062	120	7 to 533	.02 to 2.9	.05 to 1.16	64
		600 to 2000	.097		27	140	9 to 539	.10 to 4.0	.17 to 1.52	117
		800 to 2000	.059		27	229	12 to 539	.12 to 1.06	.14 to 0.94	53
3		1200 to 2000	.2		27	50,67	17 to 185	.28 to 0.53	.45 to 0.83	[a]17
4		1000	.25,.5	2.1	37	18.5,37,74	12 to 230	.18 to 1.43	.55 to 1.14	[b]80
			OD	ID	L					
4	Vertical	1000	0.875	0.375	70	42	23 to 220	$1.12 \text{ and } 1.68\times10^{6}$	$1.09 \text{ to } 1.68\times10^{6}$	16
5			.875	.54	72,108	82,123	26 to 400	.56 and 1.12	.35 to 1.12	
			.875	.375	72,108	43,65	25 to 300	.56 and 1.12	.48 to 1.36	122
			.710	.375	72	74	45 to 250	.56 and 1.12	.52 to 1.16	
6		600 to 1450	.995	.50	29,36	20,24	3 to 330	0.2,0.4,0.6,1.68,2,4,6	.69 to 1.44	
			.875	.54	102	117	16 to 270	1.12	.36 to 0.96	
			.875	.375	70,108	42,65	18 to 330	0.56, 1.112, 1.68	.66 to 1.49	[c]166
			.710	.375	70	72	14 to 170	1.68, 2.24	.73 to 1.25	
			.745	.50	36	59	9 to 115	2,4,6	.89 to 1.23	
7		1000	1.90	1.38	42	34	25 to 125	0.3 to 1.04	.49 to 0.88	27
			b	w	L					
2	Inclined 45°	2000	0.05	1.0	12.062	120	9 to 340	$0.17 \text{ to } 1.85\times10^{6}$	$0.24 \text{ to } 1.05\times10^{6}$	35
			.097	1.0	12.062	62	10 to 60	.17 to 1.31	.37 to 0.97	8
				ID						
			-------	0.187	------	67	11 to 136	.65 to 6.16	.57 to 1.57	16
			OD	ID	L					
8	Horizontal	100 to 800	1.63	1.43	42,282	97,651	51 to 344	$0.43 \text{ to } 21.7\times10^{6}$	$0.10 \text{ to } 0.68\times10^{6}$	31
9	Horizontal	600 to 800	1.81	1.43	242	288	94 to 120	.42 to 1.7	.13 to 0.38	14

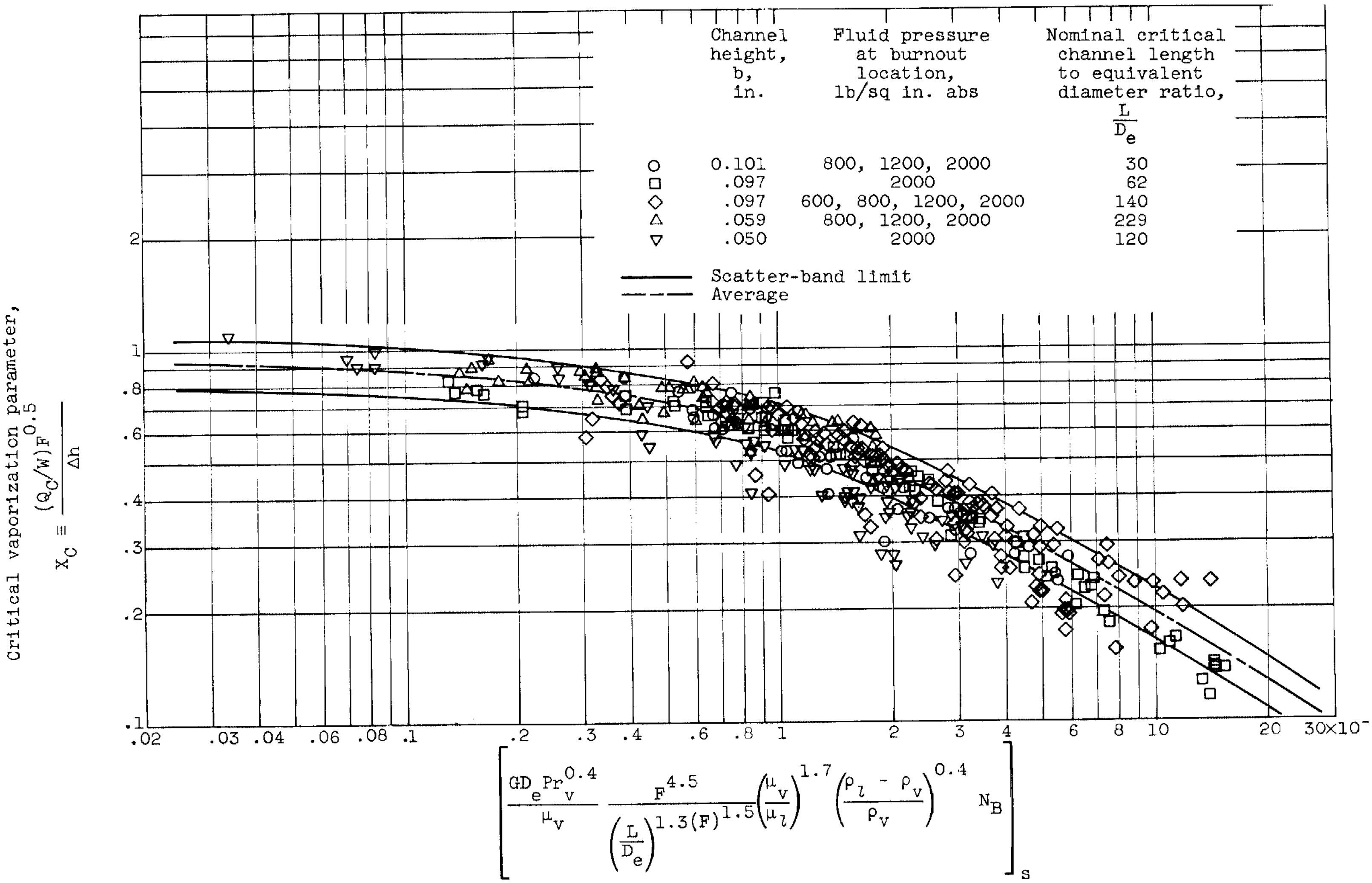
[a]Forced flow only.

[b]Valid data according to author.

[c]Only includes data with fluid quality greater than 0.01.

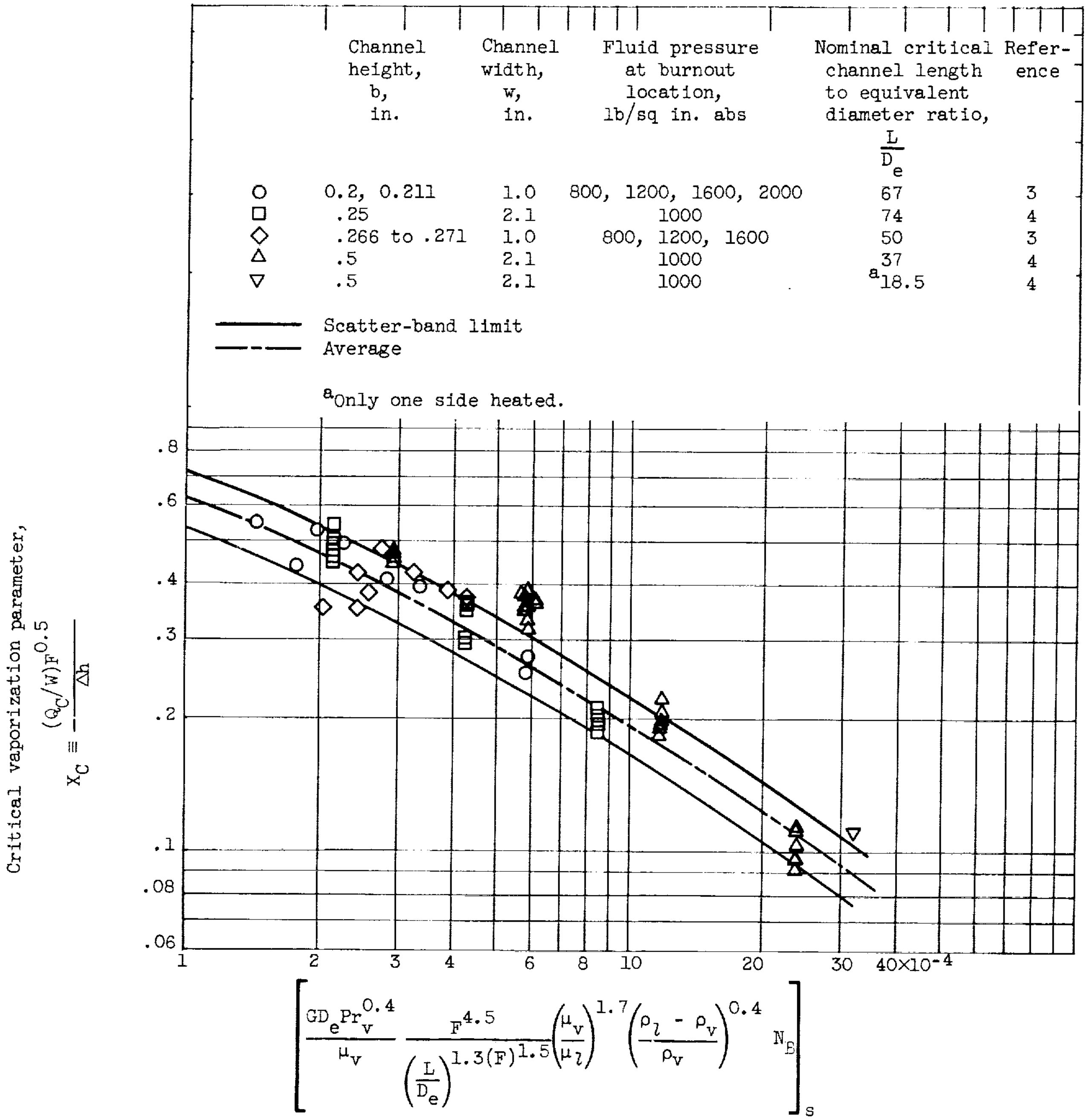
TABLE II. - CHANNEL-SHAPE FACTORS FOR VARIOUS CONFIGURATIONS

Configuration	Heated perimeter	Equivalent diameter, D_e	Effective channel height, b_e	Channel-shape factor, F
Circular tubes	Circumference	D	r	$\frac{2r}{D} = 1.0$
Rectangular channel	2 parallel sides, 2w	$\frac{4bw}{2w} = 2b$	$\frac{b}{2}$	$\frac{2b}{4b} = 0.5$
Rectangular channel	1 side heated, w	$\frac{4bw}{w} = 4b$	b	$\frac{2b}{4b} = 0.5$
Annular tube	Inner tube circumference	$\left(\frac{D_2^2 - D_1^2}{D_1}\right)$	$\left(\frac{D_2 - D_1}{2}\right)$	$\left[\frac{2\left(\frac{D_2 - D_1}{2}\right)}{\frac{\left(D_2^2 - D_1^2\right)}{D_1}}\right]^{\left(\frac{D_1}{D_2}\right)^{0.4}} = \left(\frac{D_1}{D_2 + D_1}\right)^{\left(\frac{D_1}{D_2}\right)^{0.4}}$



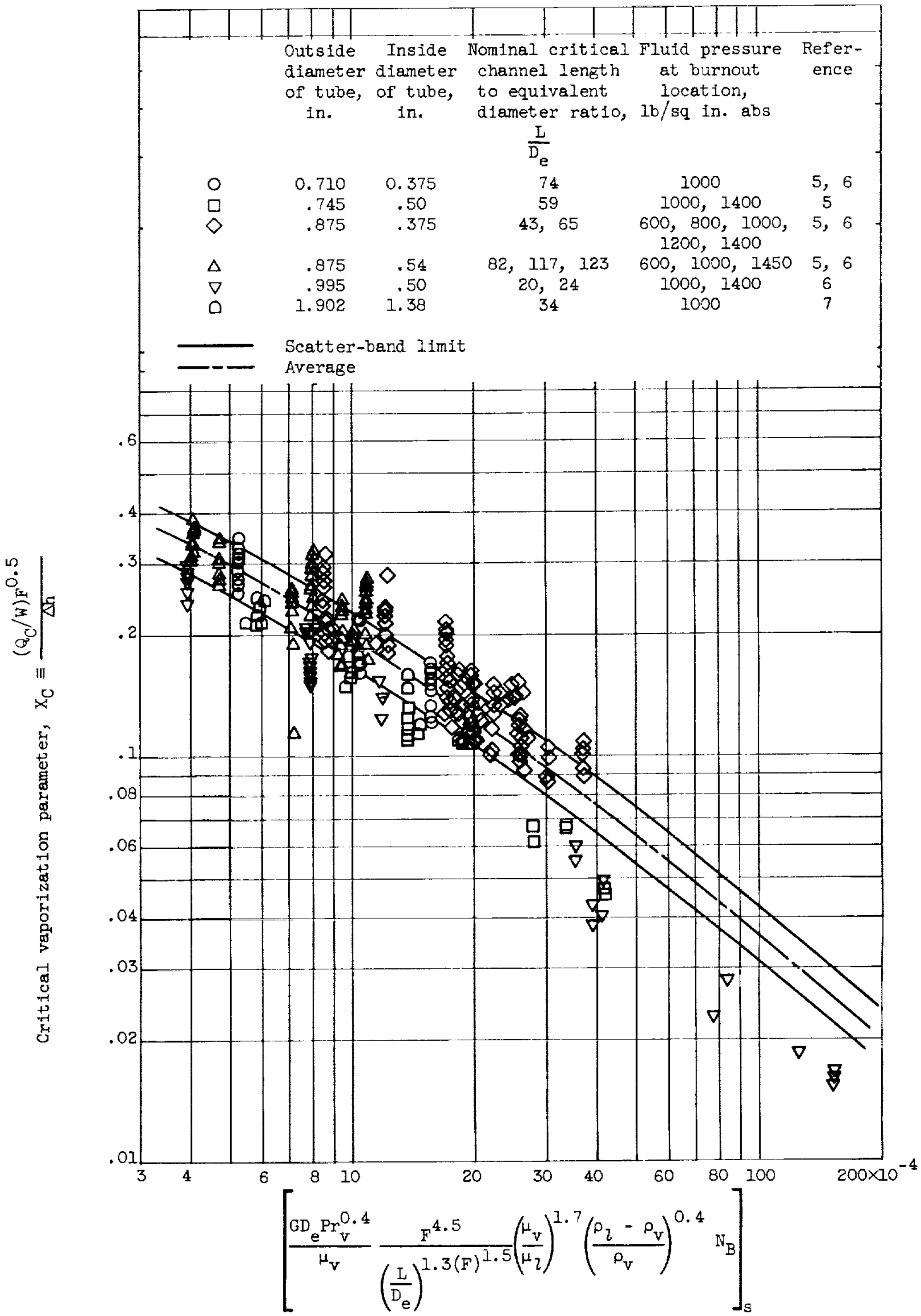
(a) Channel width, 1.0 inch (Data from ref. 2).

Figure 1. - Correlation of critical vaporization parameter for rectangular channels in terms of channel geometry, mass velocity, and fluid property parameters. Vertical flow.



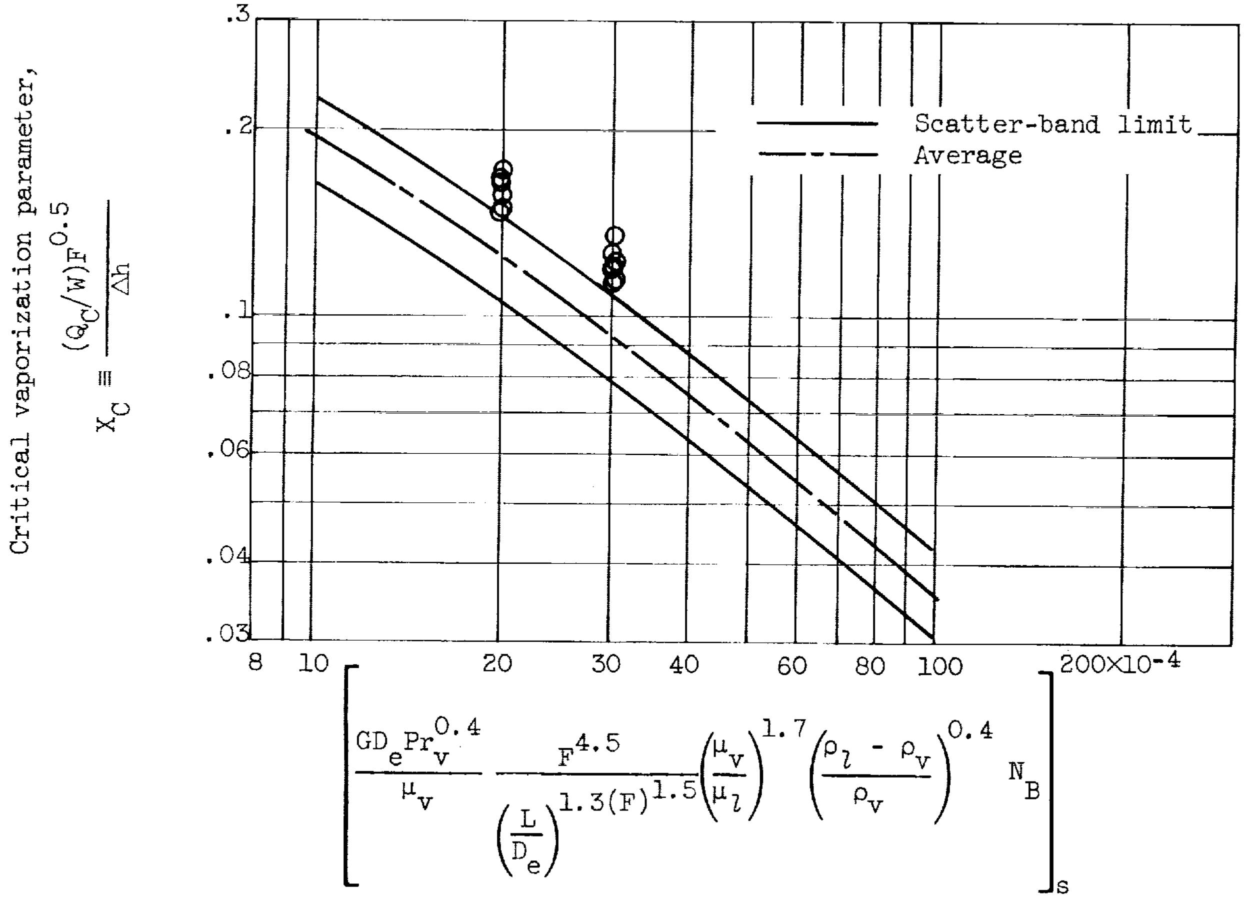
(b) Channel width, 1.0 and 2.1 inches.

Figure 1. - Concluded. Correlation of critical vaporization parameter for rectangular channels in terms of channel geometry, mass velocity, and fluid property parameters. Vertical flow.



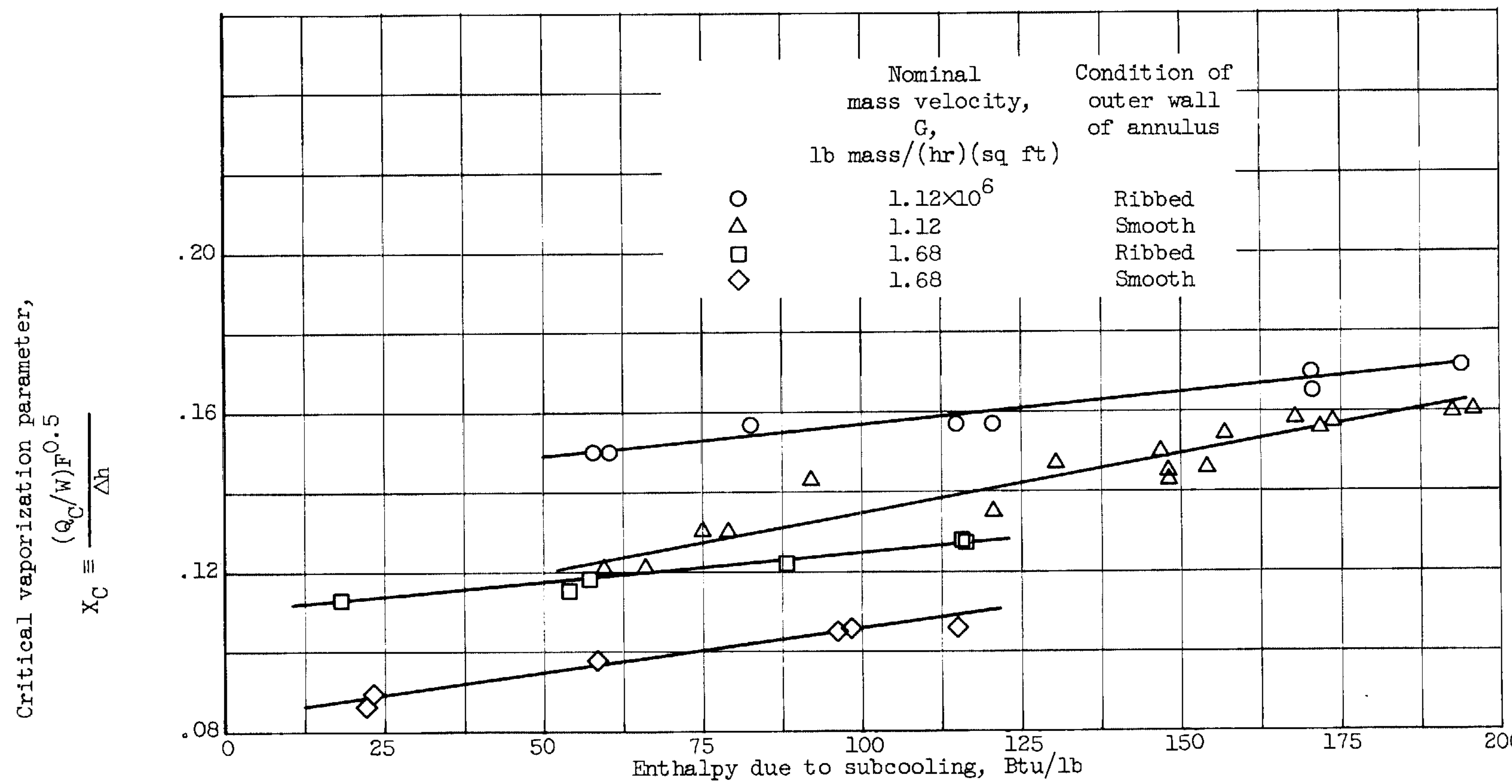
(a) Outer wall of annulus smooth.

Figure 2. - Correlation of critical vaporization parameter for annular tubes in terms of tube geometry, mass velocity, and fluid property parameters. Vertical upward flow.



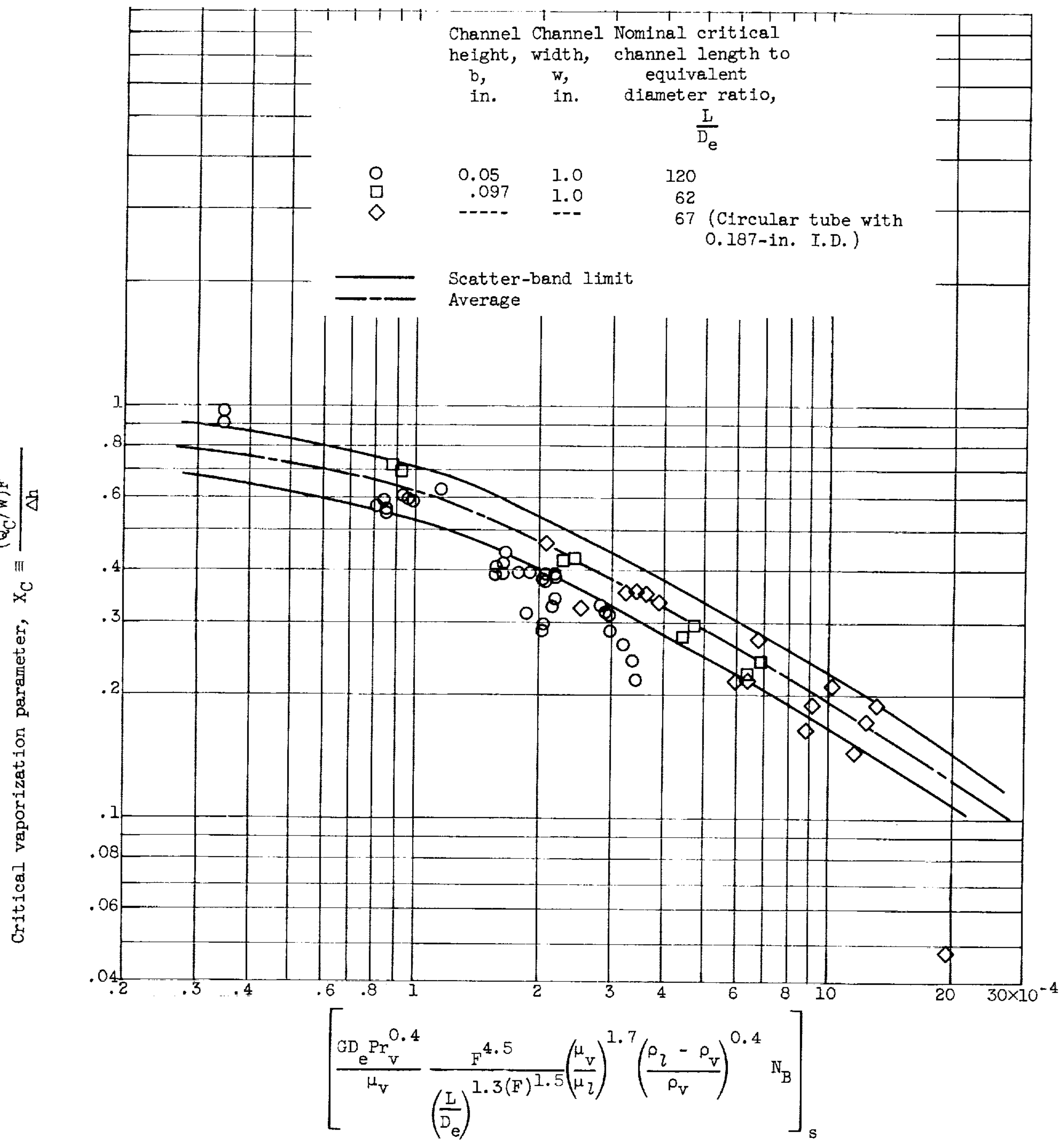
(b) Outer wall of annulus ribbed. Tube diameters, 0.875-inch outside diameter and 0.375-inch inside diameter; nominal critical channel length to equivalent diameter ratio, 42; fluid pressure at burnout location, 1000 pounds per square inch absolute (ref. 4).

Figure 2. - Continued. Correlation of critical vaporization parameter for annular tubes in terms of tube geometry, mass velocity, and fluid property parameters. Vertical upward flow.



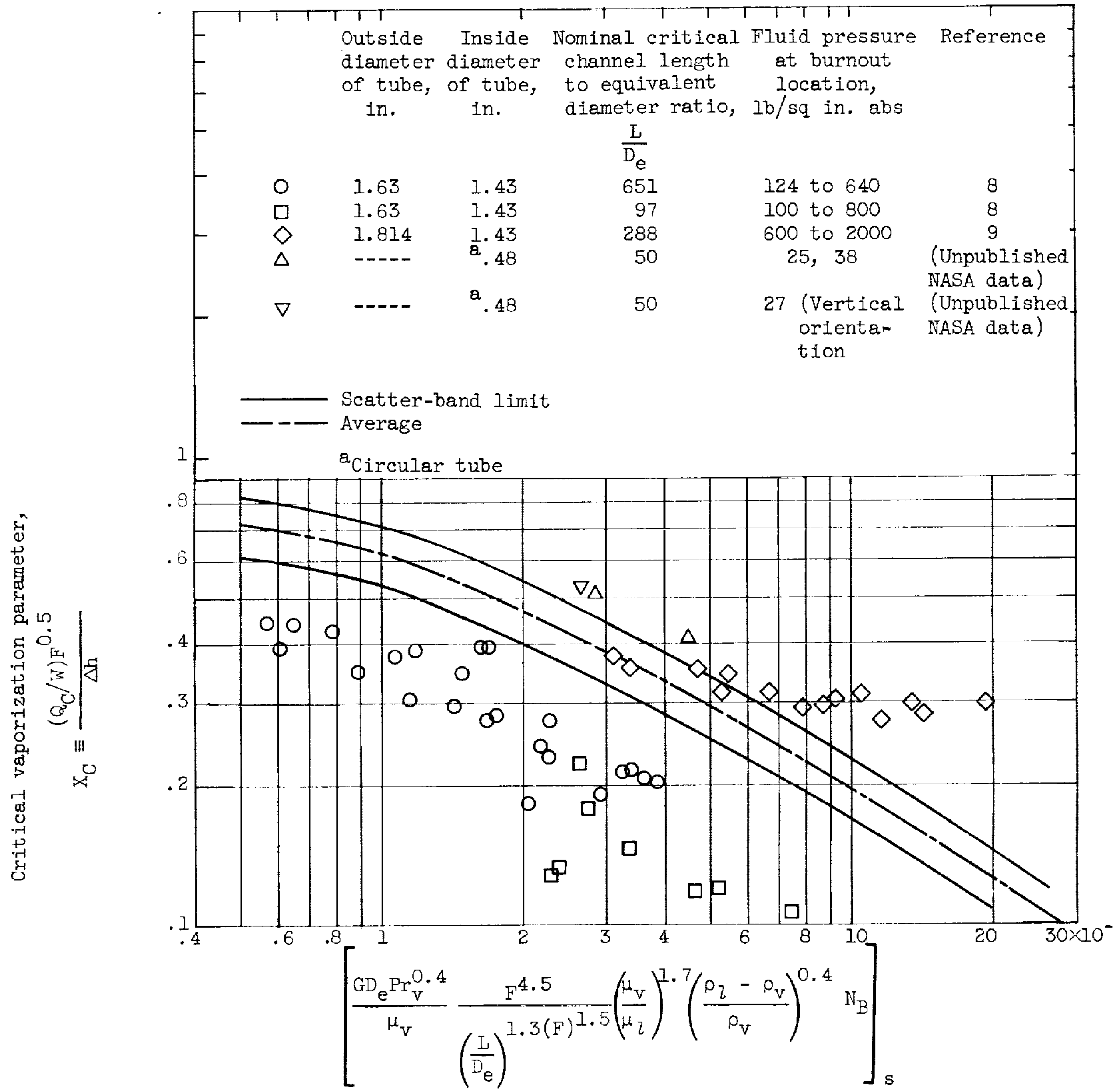
(c) Comparison of critical vaporization parameter for smooth and ribbed outer wall of annulus as function of subcooling enthalpy values. Tube diameters, 0.875-inch outside diameter and 0.375-inch inside diameter; nominal critical channel length to equivalent diameter ratio, 42; fluid pressure at burnout location, 1000 pounds per square inch absolute (ref. 4).

Figure 2. - Concluded. Correlation of critical vaporization parameter for annular tubes in terms of tube geometry, mass velocity, and fluid property parameters. Vertical upward flow.



(a) 45° inclined upward flow. Fluid pressure at burnout location, 2000 pounds per square inch absolute (ref. 2).

Figure 3. - Correlation of critical vaporization parameter in terms of channel geometry, mass velocity, and fluid property parameters. Orientation other than vertical.



(b) Horizontal channel orientation.

Figure 3. - Concluded. Correlation of critical vaporization parameter in terms of channel geometry, mass velocity, and fluid property parameters. Orientation other than vertical.